William J. Cahill, Jr.
Vice President

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February 9, 1973

Re Indian Point Unit No. 2
AEC Docket No. 50-247

Mr. R. C. DeYoung, Assistant Director
for Pressurized Water Reactors
Directorate of Licensing
U. S. Atomic Energy Commission
Washington, D. C. 20545

Dear Mr. DeYoung

We are enclosing the additional information concerning the re-analysis of safety and relief valve installations for ASME Class 1 and Class 2 systems in Indian Point Unit No. 2, requested by your letter of October 17, 1972.

The enclosed information supplements our report on this subject dated July 13, 1972.

Very truly yours

William J. Cahill, Jr.

William J. Cahill, Jr.
Vice President

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Received w/Ltr Dated 02-09-73

Additional Information Concerning the Re-analysis of Safety and Relief Valve Installations for ASME Class 1 and Class 2 Systems in Indian Point Unit No. 2

February 9, 1973

1081

QUESTION 1

Provide the dynamic load factor for open systems. Include a description of the method of analysis used to obtain this factor.

ANSWER

Thrust forces which act on the piping system upon valve operation were calculated using fluid mechanics and thermodynamic principles. The thrust forces caused by valve discharge include considerations for the fluid flow behavior and dynamic action of the valve. Equivalent static loads applied to the piping system were conservatively established by increasing the magnitude of the initial thrust force values associated with 75% of maximum flow by a factor of 2.0 to account for dynamic loading effects. The open systems at Indian Point Unit No.2 act as one degree of freedom systems. For such systems, the dynamic load factor value ranges between zero and two and has been conservatively taken as 2.0 for Indian Point Unit No. 2.

The analysis and design of the piping systems to account for valve operation has been based on a number of conservative assumptions in that the opening of the valve and the application of the blowdown loads were assumed to be instantaneous. All of the stresses resulting from the application of the thrust forces to the piping at the intersection of the branch and header were conservatively assumed to be primary stresses.

QUESTION 2

Discuss the validity of the use of ANSI B31.1 for primary plus secondary stress determination at the junction of the weldolet and the main steam header for open systems.

ANSWER

Stresses at the junction of the weldolet and the main steam header were evaluated using the CYLNOZ computer program. The CYLNOZ program is based upon Welding Research Council Bulletin No. 107, Revision 2, July 1970, entitled "Local Stresses in Spherical and Cylindrical Shells Due to External Loading" by K. R. Wichman, A. G. Hopper, and J. L. Merehon.

All of the stresses calculated using the CYLNOZ program were assumed conservatively to be primary stresses and the resulting stress values were maintained within ANSI B31.1.0 code allowables.

QUESTION 3

Submit a summary of the results of the test or analysis to verify the assumed direction of the relieving load.

ANSWER

A steam safety valve prototype was tested for three different exit nozzle configurations: 0° exit (horizontal discharge), 90° exit (vertical discharge) and 33° exit. The valve was mounted directly on a header off a pressurized drum capable of supplying a sustained steam flow at constant pressure for several seconds exceeding the time taken to achieve full accumulation flow of the valve (maximum valve lift). The tests measured the thrusts caused by the flow out of the different exhaust configurations tested, and obtained the direction of the resultant thrust force out of the 33° exit.

Strain gages mounted on a load cell forming an extension of the valve inlet nozzle section as shown in Figure 3 were used to obtain the measured strains. At the set pressure, the valve opened and continuous measurements of load cell strains, valve lift, drum pressure and load cell temperature were taken. Two strain gages read direct plus bending strains in the plane of bending, two gages read the direct strain in the plane at 90°. Four dummy gages were mounted on an identical load cell on a second steam drum header such that compensation could be achieved for any changes in temperature and pressure during the tests.

From Figure 3, the solution of the exit thrusts from measured values were obtained as follows:

$$P_o = P_V - F_V \tag{1}$$

$$M_o = F_H \times H_1 - F_V H_2 \tag{2}$$

where:

$$P_V = \frac{V^2}{v\, g_c} A$$ (mass flow force in the load cell)

Calculated values for comparison with the test values were obtained for full accumulation pressure assuming dry saturated steam. A valve efficiency factor of 0.26 (specified by valve manufacturer) was employed.

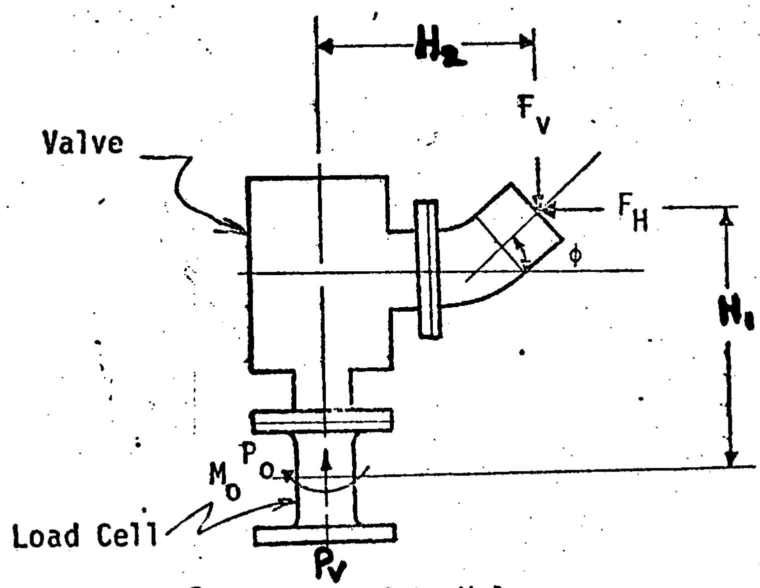


Figure 3. Steam Safety Valve

The test records show typical valve opening and valve closing times as 0.060-0.080 seconds. This causes a rapid change in exit thrust force which cause transient vibrations of the valve. The frequency of the oscillations correspond to the natural frequency of the system tested (22.7 Hz). The maximum oscillatory force component recorded by the load cell was close to 1/4 of the steady-state load component. Thus for the system tested, a dynamic load factor of 1.25 was not exceeded. No significant vibrations of the valve in the steady-state blowdown phase were observed. Good agreement between calculated and measured values were obtained. Generally, for all the exit configurations, the calculated values were conservative within 10% of the measured values.

High speed movies of the exit steam jet from the 33° angle nozzle showed a near symmetrical distribution over the exit area and did not divert significantly (less than 2°) from the normal direction to the exit plane.

QUESTION 4

Provide the following analytical parameters for each computer program referenced:

a) brief description of the theoretical basis.
b) brief description of the assumptions used for analysis and the limits of applicability.

ANSWER

The digital computer program ADLPIPE provides an elastic analysis of redundant piping systems subjected to thermal, static and dynamic loads. The system may contain a number of sections, a section being defined as a sequence of straight and/or curved members lying between two network points. A network point is: (a) a junction of two or more pipes; (b) an anchor or any point at which motion is prescribed; or (c) a position of lumped mass. A network point may be free, or one or more of its six degrees of freedom may be constrained or displaced.

ADLPIPE computes the stresses within piping systems in accordance with ANSI B31.1. Any member in the system may sustain prescribed loads or may be subject to elastic constraint in any of its six degrees of freedom. Also at any location within the system, members may be changed, masses concentrated, springs inserted, temperature conditions varied, materials and weld configurations changed, and body forces altered.

ADLPIPE computes at each point within the piping system the forces, moments, translations, and rotations which result from the imposed anchor or junction loads, thermal gradients in the system, gravitational loads in any combination of the three orthogonal axes, wind loads, and earthquake disturbances. For seismic effects, a normal mode analysis is performed using three dimensional response spectra. The resultant internal forces and moments are computed from the root mean square of the modal forces and moments.

MEL-40 is a computer program using tensor analysis methods to analyze the flexibility of multiple branch and closed-loop piping systems subject to pressure variations, temperature variations, anchor movements, weight, and/or other prescribed loading conditions. All computations are performed in accordance with the requirements of ANSI B31.1.

The maximum problem size using the static analysis approach is 99 branches, 99 balance points, and/or 999 data points. Each data point may describe one to three elements.

The CYLNOZ computer program is based upon Welding Research Council Bulletin No. 107, Revision 2, July 1970. The computer program is designed to compute stress intensities at eight different locations due to six different external loads.

The graphical data published in Welding Research Council Bulletin No. 107 are stored as data in tabular fashion.

AITKEN's method for Lagrange interpolation at unequal interval technique is employed in performing interpolation along the curve while linear interpolation is performed between the curves and thus appropriate values of non-dimensional parameters against the non-dimensional variable (β) and gamma (γ) are obtained.

The FLASH-IV program was used to calculate the hydraulic loads applied to piping systems. A description of the program is presented in WAPD-TM 80, "FLASH-IV: A Fully Implicit Fortran IV Program for the Digital Simulation of Transients in a Reactor Plant", T. A. Porsching, J. H. Murphy, J. A. Redfield, V. C. Davis.

Significant assumptions used in the analyses were presented in Applicant's report entitled, "Summary Report of Safety and Relief Valve Installation and Re-Analysis, dated July 13, 1972".

QUESTION 5

Provide a brief description of the following for typical open and closed systems:

a) mathematical model

ANSWER

Thrust loading effects on piping systems were evaluated using equivalent static loads. The mathematical models used to perform these analyses represented accurately the flexibility properties of the piping system. The MEL-40 computer program described in the response to Question 4 was used to evaluate the responses of open and closed systems to the applied equivalent static loads.

QUESTION 5

Provide a brief description of the following for typical open and closed systems:

(b) input forcing functions.

ANSWER

(1) Open System

The reaction force of a safety valve used in an open system is a maximum when full flow is being discharged through the valve. The full flow reaction force should be used when designing a valve installation. Since each safety valve can open at a different time, and since all valves on a system may not be required to operate during all transients of interest, several possible combinations of forces can exist for a particular installation.

The hydraulic reaction force for a blowing steam safety valve should be based on the appropriate safety valve flow rate. Based on ASME B&PV Code, Section III, 1971, the rated flow of a safety valve is no more than 90 percent of the calculated flow relieving capacity. For the design of a safety valve installation, the expected, or maximum, flow capacity must be considered when establishing the maximum reaction force due to the blowing valve. The maximum flow capacity can be determined from Napier's formula.

$$W = 51.5\ AP$$

where:

W = Safety valve saturated steam flow, lb/hr

A = Safety valve nozzle throat area, square inches

P = Absolute pressure at safety valve inlet nozzle, psia

The pressure to be applied in this formula is 110 percent of the design pressure of the system on which the safety valve is being installed. This is because the maximum allowable overpressure under the code to which the system is designed is 110 percent of design pressure. The nozzle throat area used is obtained from the valve manufacturer.

The total hydraulic reaction force for a discharging jet of fluid is comprised of a pressure-area contribution and a fluid momentum contribution where both of these quantities are referenced to fluid conditions at the outlet plane of the flow geometry. The total steady-state hydraulic force at the outlet may be expressed as the sum of the pressure and momentum forces as follows:

$$\frac{F}{A_o} = 144\ (P_o - 14.7) + \frac{V_o^2}{v_o\ g_c}$$

where:

F = total hydraulic force at jet outlet plane (lbs)

A_o = outlet flow area (ft^2)

P_o = outlet pressure (psia)

V_o = outlet fluid velocity (ft/sec)

v_o = outlet fluid specific volume (ft^3/lb_m)

g_c = 32.2 lb_m/lb_f (ft/sec^2)

All of the above fluid conditions are referenced to the outlet plane of the jet. Direction of force is opposite to that of discharging jet flow.

The valve isentropic expansion efficiency (n) must also be considered in the force calculation. The efficiency is not known precisely and may vary depending on the valve configuration and the operating parameters. The effect of variations in this efficiency on the reaction force calculated for a particular valve configuration is shown in the attached Figure 5b. In this case, the reaction force generally increased as isentropic efficiency increased; for other valves having different ratios of nozzle to outlet areas, the curve may differ from that shown. In order to establish the most conservative reaction force, the calculation described herein should be performed over a range of isentropic expansion efficiencies from 0.25 to 0.50.

The force calculation requires that the outlet pressure and fluid conditions be known. The calculation procedure to determine the outlet conditions and reaction force is as follows:

(1) Establish the conditions at the valve inlet

$$P,\ h,\ S$$

where:

h = saturated steam enthalpy at P (Btu/lb)

S = saturated steam entropy at P (Btu/lb F)

(2) Determine the valve flow for these conditions as described above, i.e.,

$$W = 51.5\ AP$$

(3) Assume an outlet pressure P_o; determine an isentropic exit enthalpy, h_s, based on the assumed outlet pressure P_o and the inlet entropy S

$$h_s = h_s\ (P_o,\ S)$$

(4) For a value of isentropic expansion efficiency (n) within the range from 0.25 to 0.50, calculate the outlet enthalpy h_o for the assumed outlet pressure.

$$h_o = h - n(h - h_s)$$

(5) Calculate the outlet velocity based on the expression:

$$V_o = 223.8\ \sqrt{h - h_o}$$

(6) Determine the outlet specific volume v_o based on the assumed outlet pressure P_o and the outlet enthalpy h_o.

(7) With the outlet conditions established above, calculate the mass flow rate by:

$$W = \frac{A_o\ V_o}{v_o}$$

(8) If the mass flow rate calculated above is the same as the maximum flow capacity of the valve, (from $W = 51.5AP$), then the assumed outlet pressure and calculated outlet conditions are compatible with the inlet conditions and the reaction force can be compatible with the inlet conditions and the reaction force can be calculated. If the mass flow rate calculated does not agree, then a new outlet pressure must be assumed and the procedure repeated.

(9) Repeat the above calculation for other values of isentropic expansion efficiency within the specified range, and select the highest value of reaction force for the design of the safety valve installation.

(2) Closed System

In a closed system, no sustained reaction force from a free discharging jet of fluid can exist. However, transient hydraulic forces can be imposed at various points in the piping system from the time a safety valve begins to pop open until steady flow is completely developed. In order to evaluate these transient forces, an analytical model of the system being analyzed must be developed. In developing models of pressurizer safety valve systems, Westinghouse has applied technology developed for the analysis of the blowdown during a loss-of-coolant accident. The analytical models developed and their applications are considered to conservatively estimate the transient forces.

The FLASH-IV digital computer code (Reference 1) has been used to calculate the transient hydraulic loads in closed piping systems. For these calculations, the system is modeled by dividing the piping system into control volumes and defining the control volume characteristics required by the FLASH-IV program. Valves are modeled as "leak elements" with choked flow into the valve calculated by the Moody correlation. The code is also set up to check flow choking at the end of the discharge piping. Frictional losses are incorporated for the discharge piping and associated elbows.

The following assumptions have been incorporated into the calculation of transient hydraulic loads on closed safety valve piping systems with a water seal on the upstream side of the safety valve.

1. Valve Opens Full in 40 Milliseconds

   Information from valve manufacture indicates opening to approximately 70% in 40 milliseconds.

2. Loop Seal Water is Pushed Ahead of Steam

   Actually some breakup of the water mass is expected to occur as water is forced past the valve seat and as the water passes through successive downstream elbows.

3. Two-Phase Flow in the Downstream Piping is Homogeneous

   Thus, any flashing of loop seal water will result in steam bubbles trapped in the water mass. Actually some phase separation is likely, thereby reducing the acceleration of the liquid phase.

4. No Credit is Taken for Power-Operated Relief Valves

Actuation of power-operated relief valves would increase the back-pressure in the relief tank piping system, thereby reducing the transient hydraulic loads from subsequent safety valve actuation. For this analysis, the lowest back-pressure was assumed (3 psig) corresponding to conditions just prior to actuation of the first safety valve with relief valves closed.

Reference:

1. WAPD-TM-80, "Flash-IV: A Fully Implicit Fortran IV Program for the Digital Simulation of Transients in a Reactor Plant", T. A. Porsching, J. H. Murphy, J. A. Redfield, V. C. Davis.

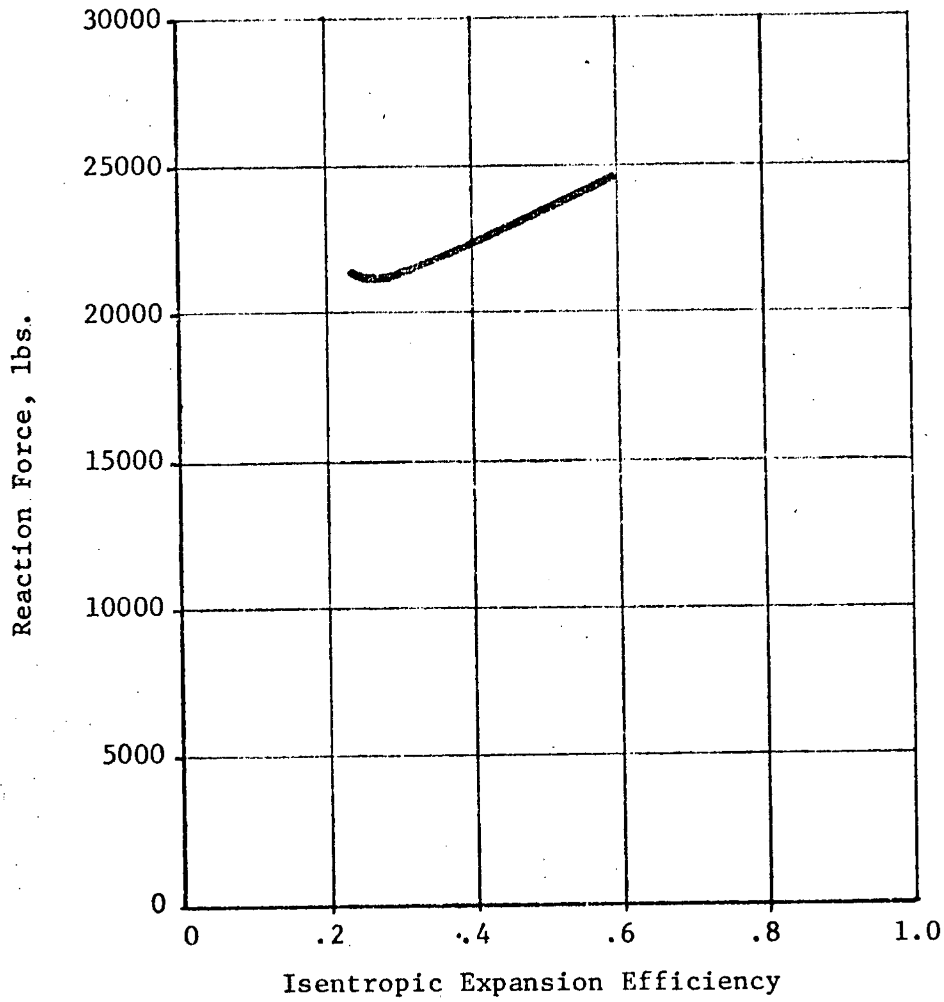


FIGURE 5b

VARIATION OF VALVE REACTION FORCE

WITH ISENTROPIC EXPANSION EFFICIENCY

QUESTION 5

Provide a brief description of the following for typical open and closed systems:

c) summary of stresses at high changes in flexibility.

ANSWER

The stresses at each point in the piping systems have been examined for the highest combination load and have been found generally to be well within allowable limits. The points of maximum stress have been found to exist at elbows and tees (i.e., high changes in flexibility) or at support points. These maximum stress values have been tabulated in Table 3 of the July 13, 1972 report entitled, "Summary Report of Safety and Relief Valve Installation and Re-Analysis for ASME Class 1 and Class 2 Systems in Indian Point Unit 2".

QUESTION 6

Provide justification for using a 0.15g horizontal and a 0.10g vertical DBE input in lieu of the results from a multi degree-of-freedom system.

ANSWER

The ground accelerations of 0.15g horizontal and 0.10g vertical input have been established in the Indian Point Unit No. 2 FSAR as the Design Basis Earthquake. However, these values were not used in the seismic analysis of the piping systems. Piping associated with the pressurizer and main steam relief valve systems was analyzed for seismic effects using the ADLPIPE computer program. The dynamic analysis incorporated multi-degree-of-freedom mathematical models and appropriate response spectra curves. Seismic stresses for the other piping systems containing relief valves were determined using the MEL-40 program. Equivalent static loads were calculated using acceleration coefficients associated with the peak values of the ground response spectra curves with appropriate damping factors.

QUESTION 7

Provide justification for the use of a load factor of 2 for closed systems.

ANSWER

Based on the characteristics of the transient hydraulic loads from the valve opening in closed steam systems, the maximum dynamic load factor of single-degree-of-freedom systems to similar transients, and the dynamic load factor based on time history analyses of a number of closed systems in a typical pressurized water reactor plant, a dynamic load factor of two is conservative.

In a closed steam system (no loop seal), the transient hydraulic loads acting on each leg of the downstream piping as calculated by FLASH IV characteristically have one major pulse which is one-sided (all plus). A typical transient hydraulic force is shown in Figure 7-1. Such transient hydraulic forces excite each leg in turn, as the pressure pulse travels downstream so that each leg experiences one pulse. Even though the piping system as a whole experiences a series of pulses.

The dynamic load factor of undamped single-degree-of-freedom systems for a number of different single-sided transients is well known(1). The dynamic load factors for two common transients are shown in Figures 7-2 and 7-3. The maximum dynamic load factor for a single-degree-of-freedom system for a single, one-sided pulse is two.

However, for a symmetrical two-sided pulse, the maximum dynamic load factor for an undamped single-degree-of-freedom system can be greater than two as shown in Figure 7-4. For example, the maximum dynamic load factor for a two-sided triangular pulse is less than 2.7. Therefore, the two-sided transient represents a more severe loading condition than the single-sided transient.

The theoretical results are all for undamped systems. The dynamic load factor will be smaller if damping is considered. In order to evaluate the effects of the local application of the load, the multiple degrees-of-freedom, the three-directional motion, small amounts of damping, (less than 2%), and the characteristics of hydraulic loads computed for actual piping systems, the dynamic load factor for a number of typical closed systems was determined.

The time-histories of the hydraulic forces were determined by means of the FLASH IV computer program. The resulting transient forces were two-sided transients because loop seals were included. The time-histories of the piping stresses were determined by means of the WESTDYN-FIXFM-WESDYN2 package of computer programs. The structural model of all of these systems included the pressurizer, upstream piping (including loop seal), valve, downstream piping, and the associated hangers and snubbers. Two of the systems contained six legs of downstream piping, and two contained eight legs.

The lowest fundamental frequency was 10.2 Hz, and the highest was 23.2 Hz. The dynamic load factor was determined as the ratio of the maximum stress from the time-history analysis to the maximum stress from a static load analysis.

The load used in the static analysis was chosen in several different ways. First, a true time slice of the time-histories of the hydraulic loads was taken at those times when the load could cause a maximum stress condition. Such times could be when the load on an individual leg peaked (plus or minus) or when the superposition in time of some combination of the loads peaked. At least fifteen different true time slices were evaluated for each system. From all of these time slices, the maximum stress calculated for each system is shown in Table 7-1 under the heading of True Time Slice.

Second, the peak load on each individual leg was chosen, and all of the peaks applied to their corresponding legs as one static load condition. The resulting maximum stress is shown in Table 7-1 under the heading of $\Sigma F_{i\ max}$.

Third, the peak load on each leg was applied individually, and the maximum stress from this set of loads is shown in Table 7-1 under the heading $F_{i\ max}$.

All of the actual dynamic load factors determined for these systems are less than two as shown in Table 7-1, even though the transients were more severe than the one-sided transients in closed steam systems. Since these systems include those with relatively low maximum stresses and those with relatively high maximum stresses, a design dynamic load factor of two for closed steam systems is sufficiently conservative to be consistent with current design practice.

Reference

1. Ayre, R. S. "Transient Response to Step and Pulse Functions", Chap. 8 of Shock and Vibration Handbook, C. M. Harris and C. E. Crede, Editors, McGraw Hill, 1961.

TABLE 7-1 ACTUAL DYNAMIC LOAD FACTOR FOR TYPICAL CLOSED SAFETY VALVE INSTALLATIONS IN A PWR PLANT

		Time History	True Time Slice	ΣF_i max	F_i max
1st Sys.	Stress, psi	43,700	29,000	29,400	27,500
f_1 = 10.2 Hz, f_2 = 15.4 Hz	ACTUAL D.L.F.	—	1.51	1.49	1.59
f_3 = 23.2 Hz	% SAFETY FACTOR BASED ON D.L.F.=2	—	132	134	126
2d Sys.	Stress, psi	2,600	1,800	2,100	1,900
f_1 = 23.2 Hz, f_2 = 23.2 Hz	Actual D.L.F.	—	1.45	1.24	1.39
f_3 = 23.5 Hz	% SAFETY FACTOR BASED ON D.L.F.=2	—	138	161	144
3rd Sys.	Stress, psi	41,200	25,100	31,200	27,600
f_1 = 11.9 Hz, f_2 = 19.6 Hz	Actual D.L.F.	—	1.64	1.32	1.49
f_3 = 29.8 Hz	% SAFETY FACTOR BASED ON D.L.F.=2	—	122	152	134
4th Sys.	Stress, psi	2,400	2,000	1,900	1,600
f_1 = 16.8 Hz, f_2 = 29.7 Hz	Actual D.L.F.	—	1.20	1.26	1.50
f_3 = 33.4 Hz	% SAFETY FACTOR BASED ON D.L.F.=2	—	166	158	133

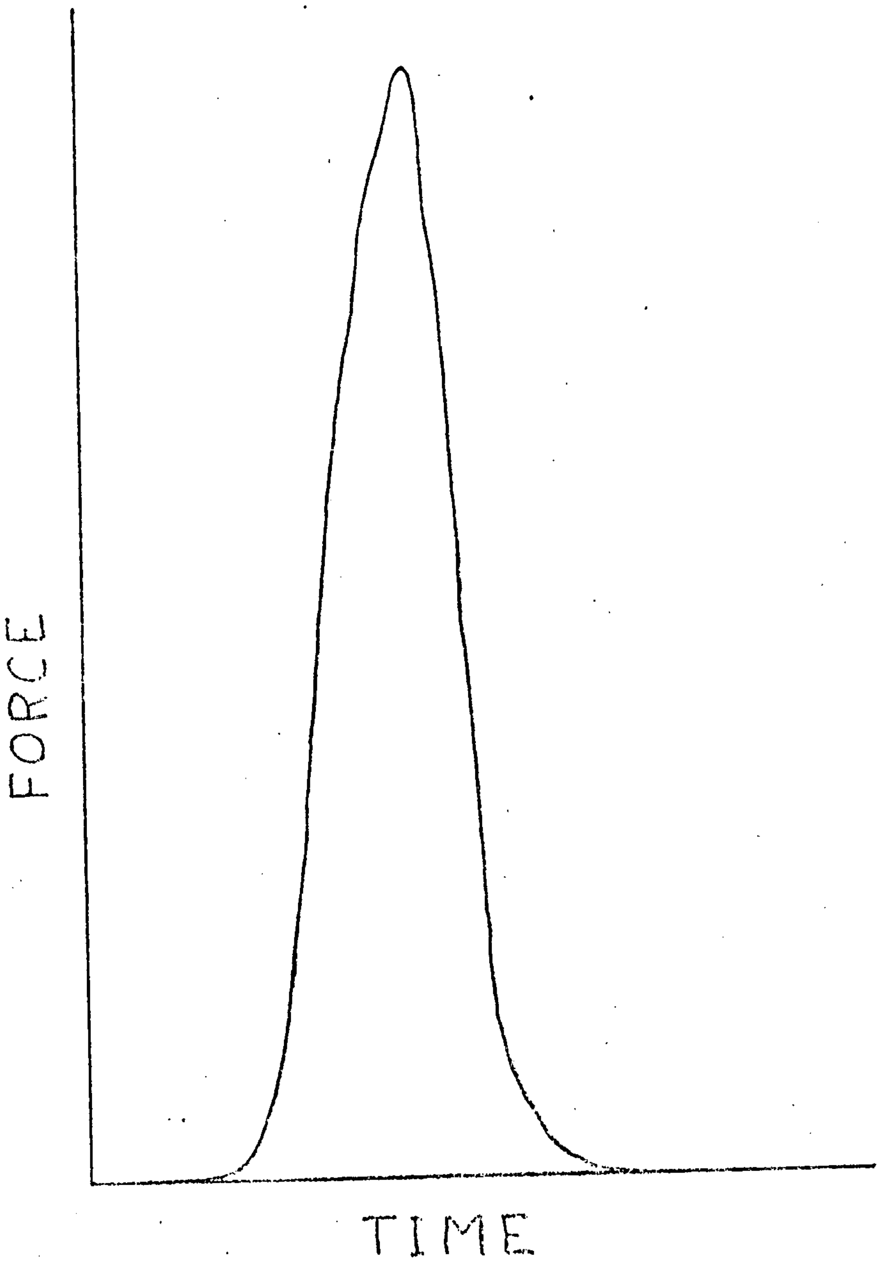


FIG. 7-1 Typical hydraulic transient for valve opening on steam in a closed system as computed by Flash IV

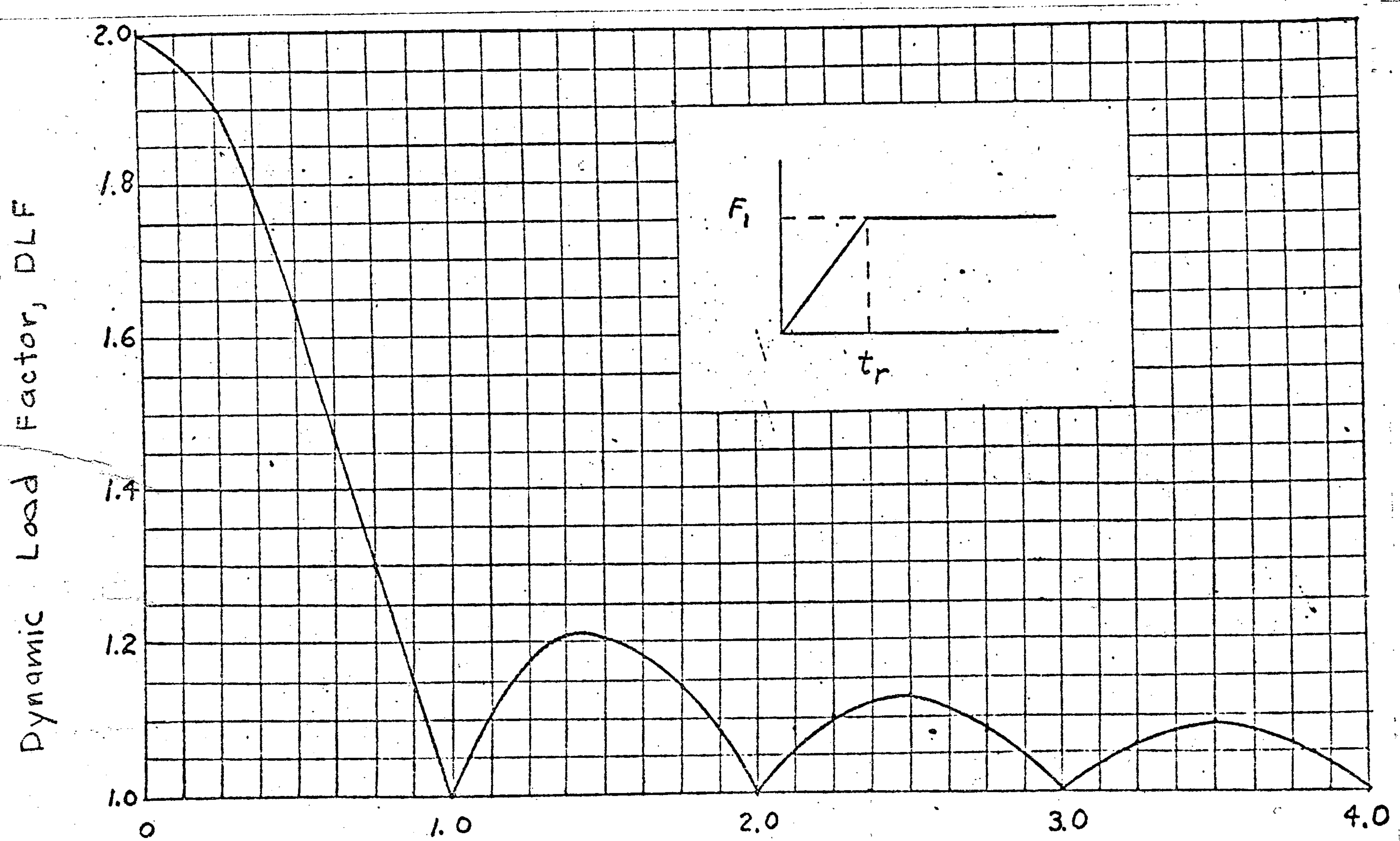
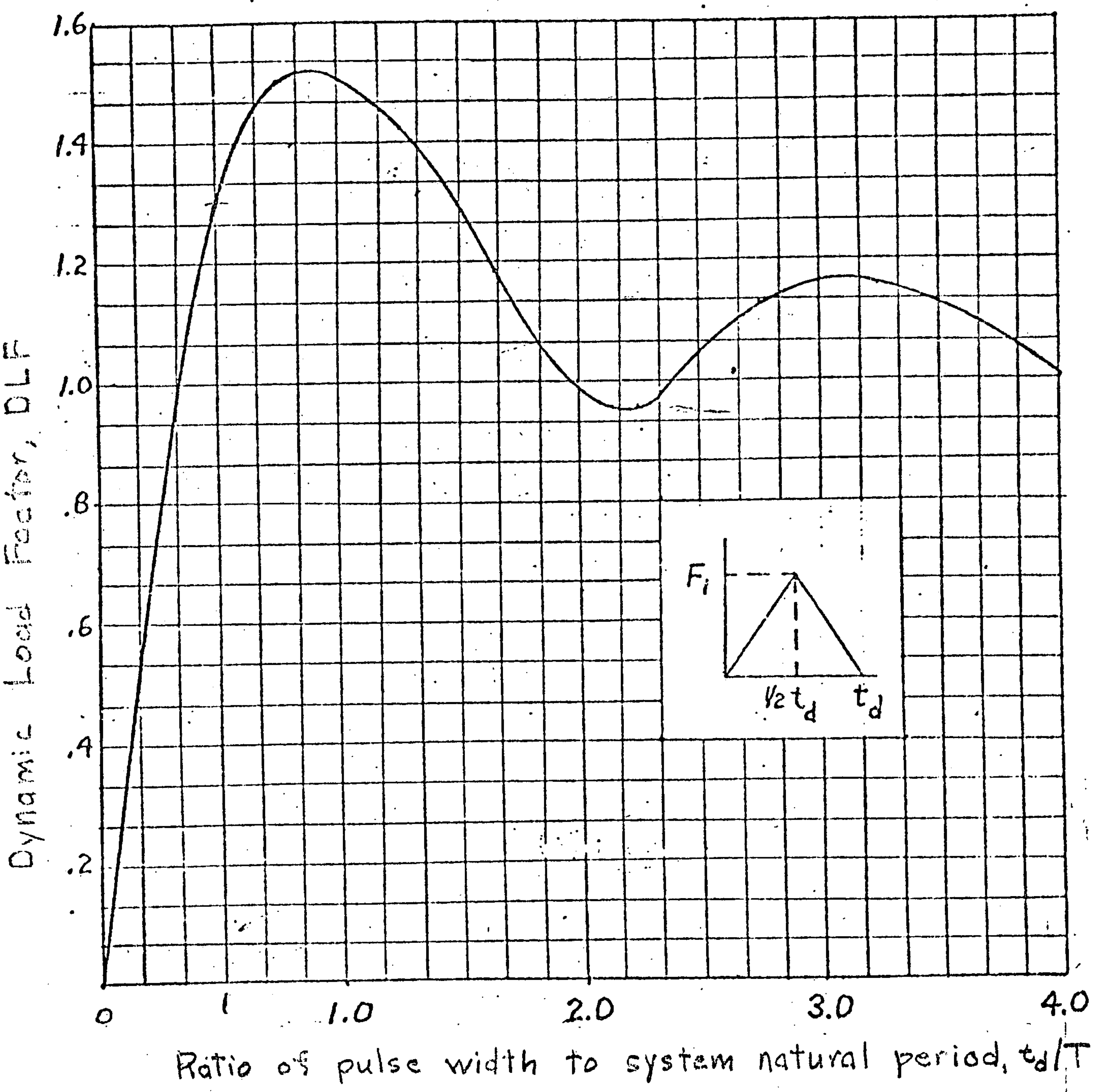


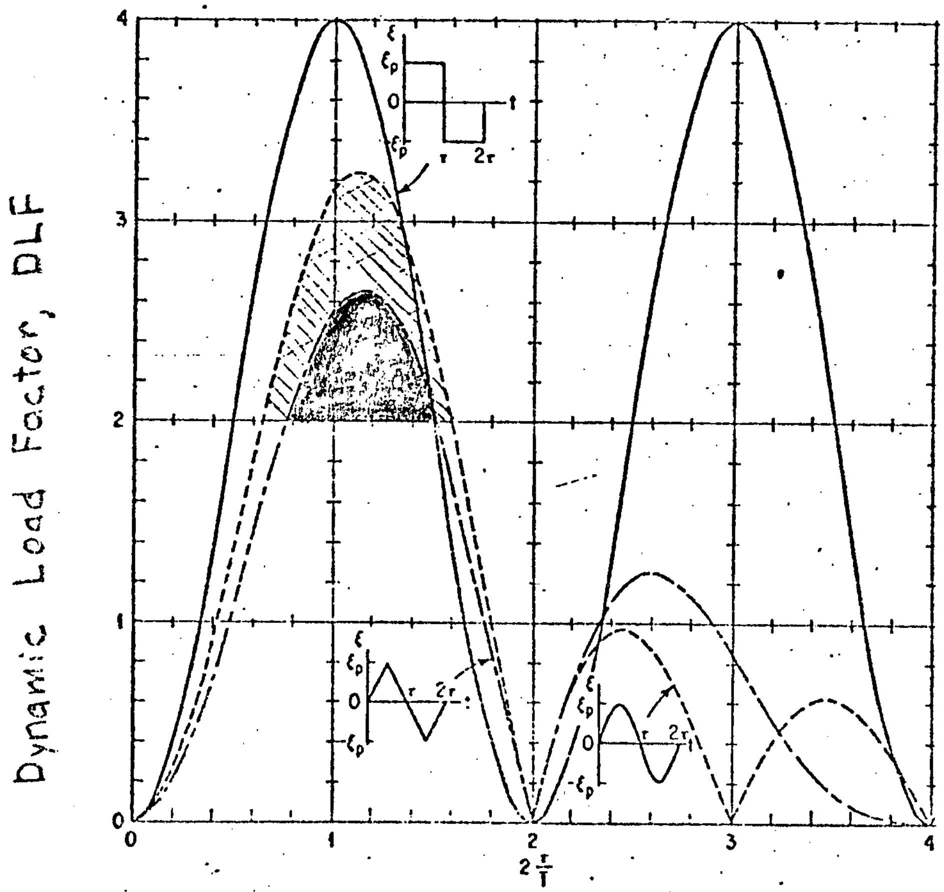
FIG. 7-2 Ratio of rise time to system natural period, t_r/T.

Dynamic Load Factor of single-degree-of-freedom elastic systems subjected to a constant force pulse with finite rise time.



Dynamic Load Factor of single-degree-of-freedom elastic systems subjected to a triangular force pulse.

FIG. 7-3



Ratio of pulse width to system natural period, $2\tau/T$.

Dynamic Load Factor of single-degree-of-freedom elastic systems subjected to two-sided force pulses.

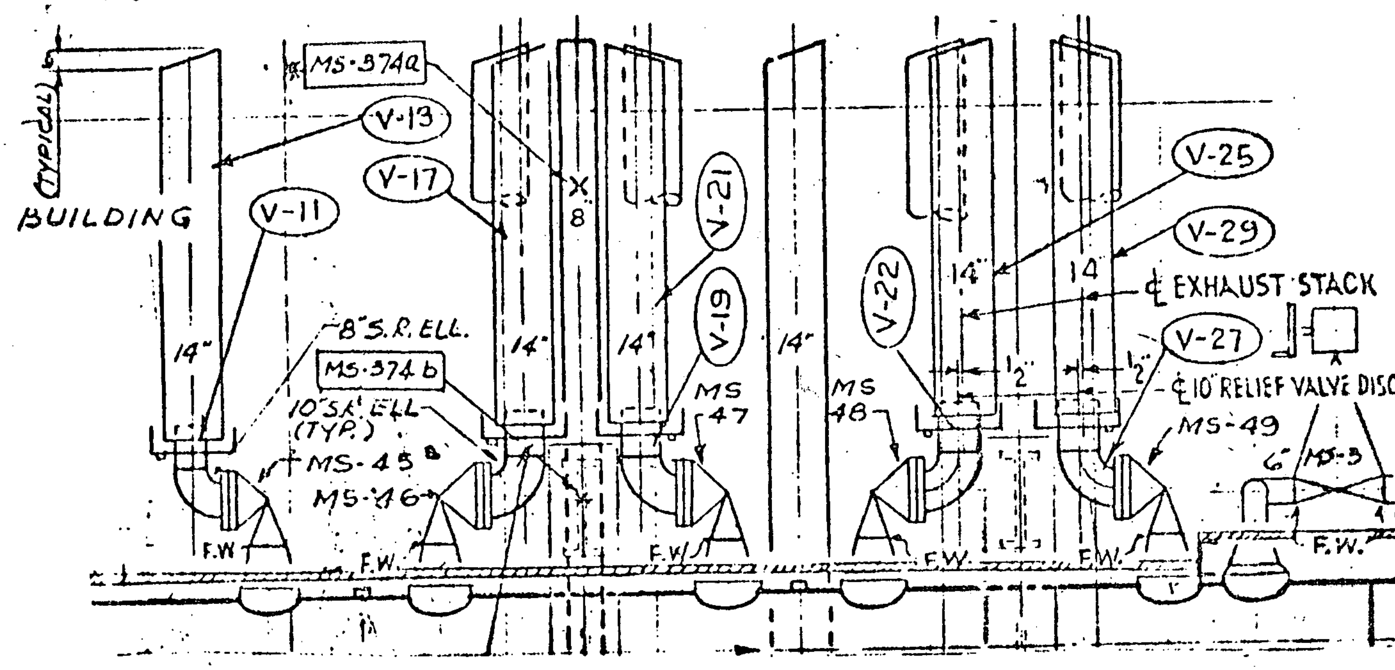
FIG. 7-4

QUESTION 8

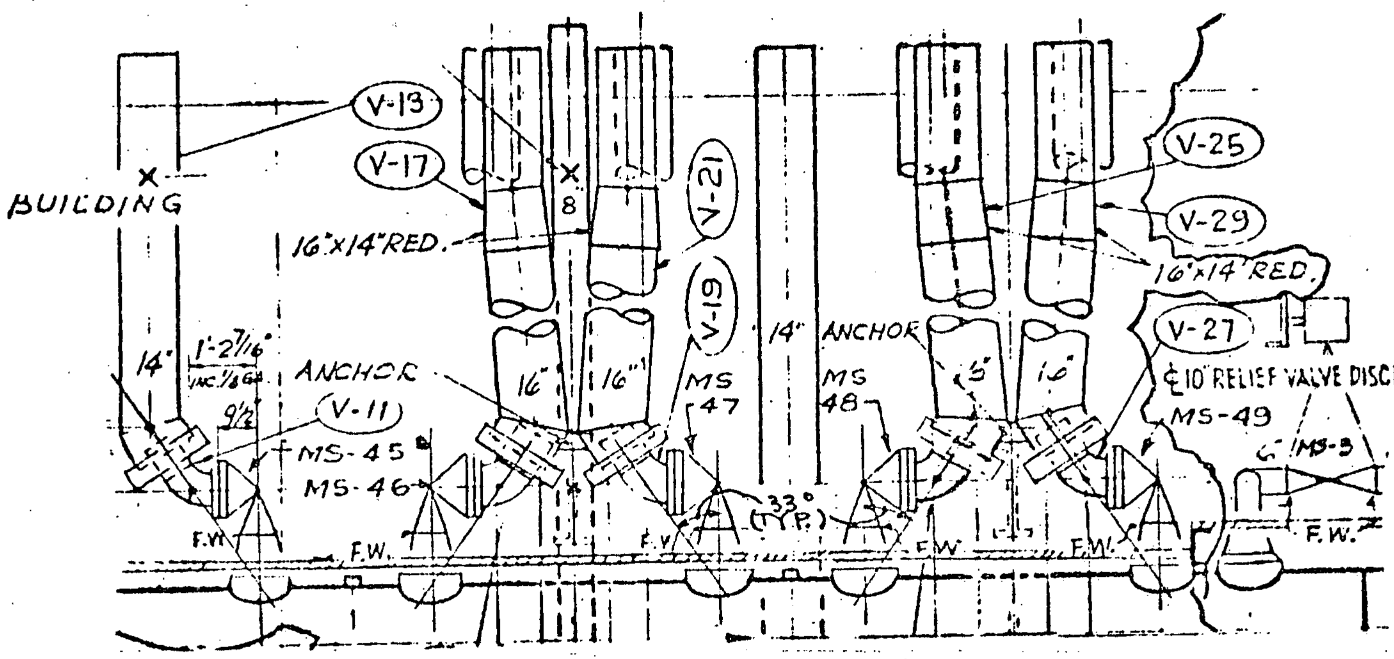
Provide sketches of the required modifications used for all typical systems.

ANSWER

The attached Figures 8a, 8b, 8c and 8d show the modifications used.



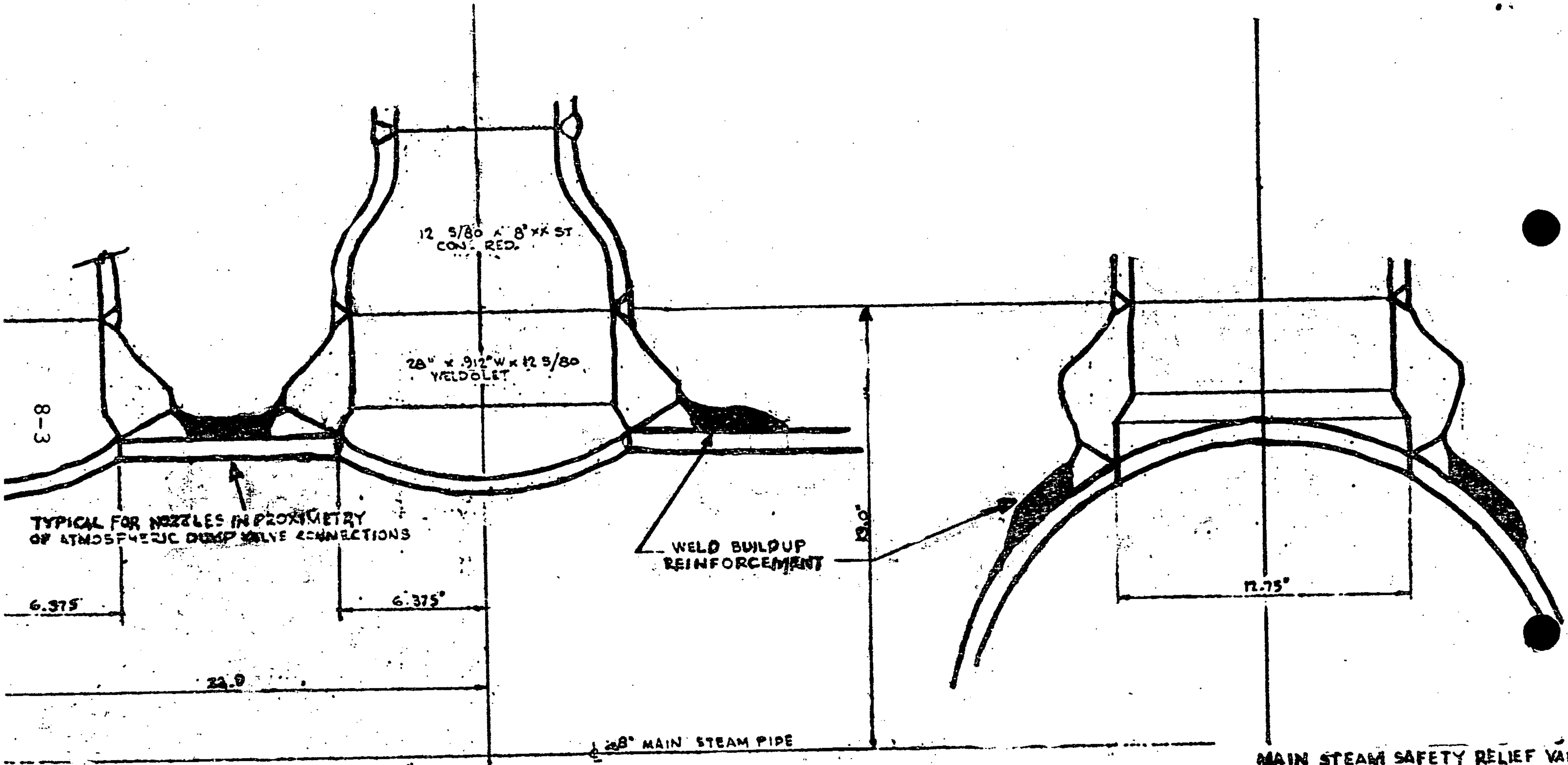
ORIGINAL CONFIGURATION



MODIFIED CONFIGURATION

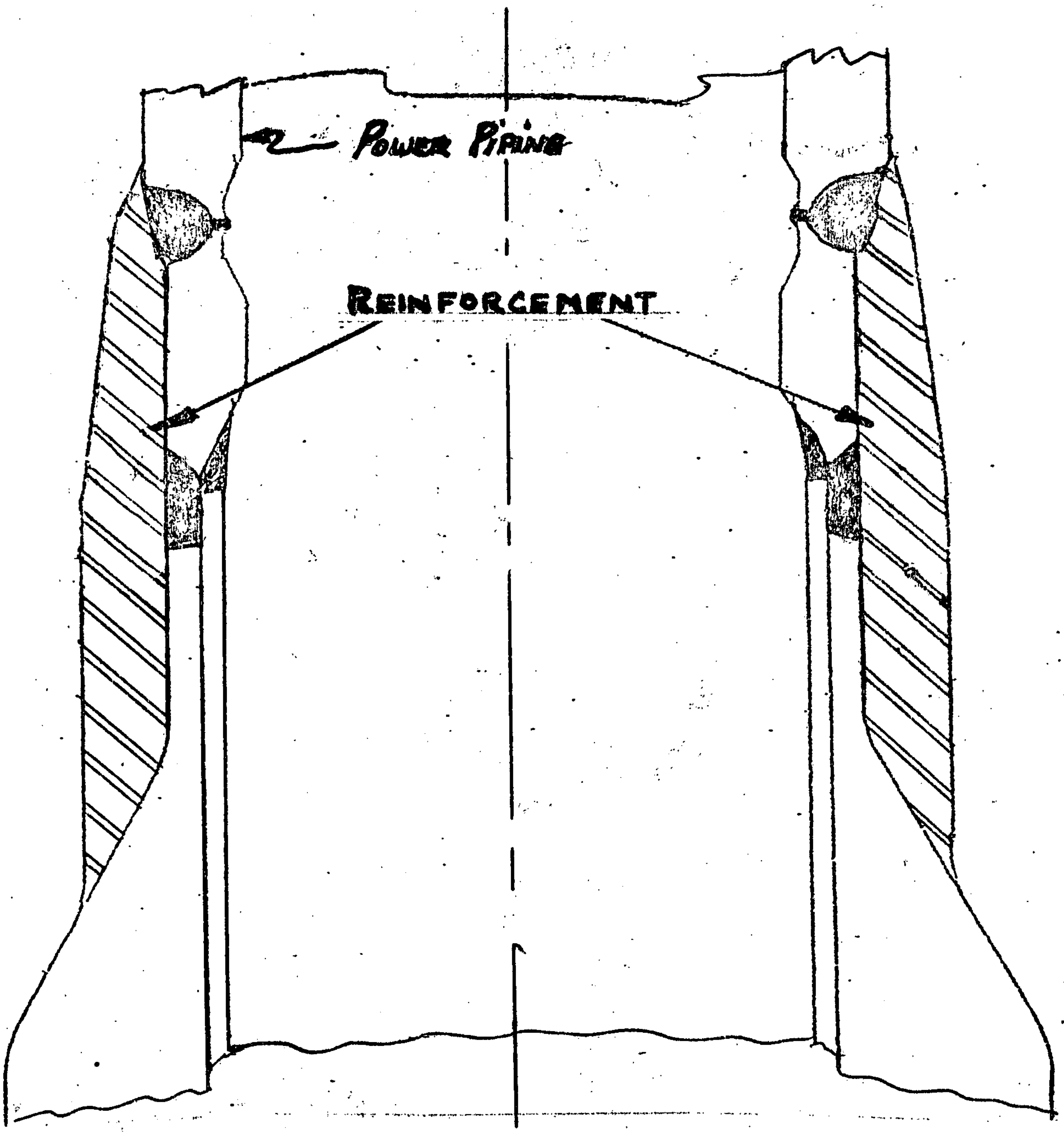
MAIN STEAM
SAFETY RELIEF VALVE
- ARRANGEMENTS -

FIG. - 8a



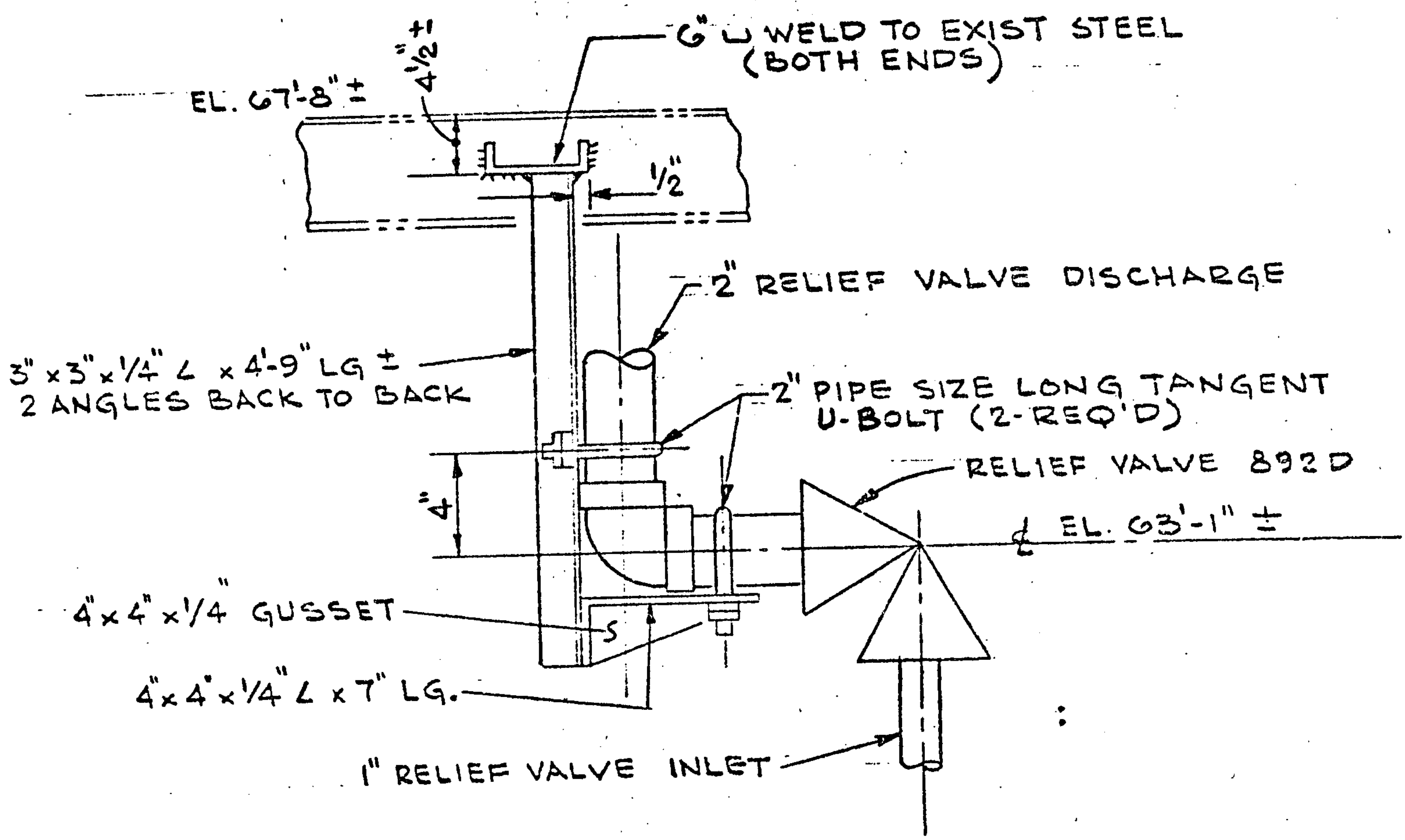
MAIN STEAM SAFETY RELIEF VALVE
NOZZLE REINFORCEMENT
— WELD BUILDUP —
INDIAN POINT UNIT #2

FIG-8b



PRESSURIZER NOZZLE

FIG-8c



AT ACCUMULATOR TANK No. 24

RESTRAINT ADDED ON A TYPICAL SYSTEM

FIG.-8d

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(TEMPORARY FORM)

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DESCRIPTION:
Ltr re their 07-13-72 submittal and our 10-17-72 ltr trans the following:

PLANT NAMES: Indian Point, Unit 2

ENCLOSURES:
Additional Info Concerning the Re-analysis of Safety and Relief Valve Installations for ASME Clas 1 & Class 2 Systems in Indian Point Unit No. 2, dtd 02-09-73.

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